PATENT SPECIFICATION

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COMPLETE SPECIFICATION

Improvements in or relating to Transmissions for Motor Vehicles

We, ZAHNRADFABRIK FRIEDRICHSHAFEN AKTIENGESELLSCHAFT, of Friedrichshafen-on-the-Bodensee, Germany, a Joint-Stock Company incorporated under German law, do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

This invention relates to an improved power transmission device for motor vehicles and consisting of a hydraulic transmission (torque converter) combined with a spur wheel change-speed gear equipped for more than two forward gear speeds and having coaxial main shafts and a countershaft parallel thereto, wherein the shaft connected to the turbine wheel of the converter is constructed as a hollow shaft and surrounds the gear input shaft, and non-positive gear shift clutches incorporated between the wheel pairs for the purpose of producing the different speeds both in individual gear stages with the use of the hydraulic transmission and in individual gear stages of the stepwise shifted gear alone while the hydraulic transmission is disconnected.

Power transmission devices hitherto disclosed for motor vehicles, having hydraulic transmissions (converters) provided for individual gear speeds, generally constitute only special constructions which usually are complicated and therefore expensive. The driving and driven shafts in known gears are in some cases offset with respect to one another. Even for simple requirements, i.e., for only a few forward speeds, known gears still contain too many individual parts. Both in extraneous power transmission devices for obtaining, for example, only two forward speed stages, and in devices for four forward speeds, a relatively large number of clutches have hitherto been provided, namely, in addition to one clutch for each of the various gear speeds, at least one separate clutch is required for bridging the pump wheel shaft and turbine wheel shaft. In some prior consructions even a larger number of clutches was provided. When use is made of planet wheel gears there are, in addition to the actual gear shift clutches also the brakes, which in this case must perform the same task as in shift clutches or other clutches. In known gears the bridging clutch is also necessary, which is generally provided as an attachment to the torque converter; on occasion a clutch of this type was incorporated at the output end of the gear.

In all constructions of gear disclosed hitherto the number of clutches used was always greater than that of the speed stages provided.

In contradistinction to the foregoing proposals, the present invention aims at simplification of the transmission by obviating any additional clutches, apart from the actual gear shift clutches. In principle, the simplification is obtained by combining with the hydraulic transmission (converter) a multi-speed step gear, which is in itself of normal consruction, while the dimensions, and particularly the structural length of the gear, remain within normal limits. Furthermore, all complicated constructions are avoided, which in addition to the purely technical advantages also affords an economic advantage.

To this end, in accordance with the present invention the gear input shaft and the turbine wheel hollow shaft are each in operative connection with one of the non-positive gear shift clutches situated inside the gear, and through the operation of which the gear input shaft is coupled in each case to the turbine wheel shaft and thus only gear shift clutches are used for connecting and disconnecting the hydraulic transmission, without additional clutches being required.

In order to establish the connection be-

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tween the pump wheel shaft and the turbine shaft, either a gear shift clutch of the countershaft or a gear shift clutch of the main gear shaft can be provided.

In order that the invention may be more readily understood, reference is made to the accompanying drawings which illustrate diagrammatically and by way of example three embodiments of transmission device according to the invention, together with diagrams depicting passage through the individual gear speeds with and without the use of the torque converter, and in which:—

Figure 1 shows a fluid gear acting as torque converter, and associated therewith a changespeed gear furnished with non-positive, preferably friction clutches in a multi-group construction, for the purpose of obtaining four forward speeds. This installation is particularly suitable for omnibuses. Associated with Figure 1 is a diagram indicating the power flow or path through the gear box in the normal gear speeds and for two further gear speeds with converter operation;

Figure 2 shows the lines of tractive force of such a gear and the resulting torque increase for selected speeds of the transmission;

Figure 3 shows another embodiment of transmission and the appertaining diagram shows the power flow or path through the various gear speeds with and without the use of the torque converter; and

Figure 4 is modification of the embodiment of Fig. 1 with its appertaining diagram.

Referring to Figure 1, the engine shaft M has keyed on to it the pump wheel P of a fluid gear constituting a torque converter and a wheel 1 (on the engine side) of a pair of wheels 1,2 for transmission without the converter. Following the wheel 1 is a shift clutch 4, 5, one part 5 of which is mounted on a shaft part 7, which also carries a fixed wheel 8 and one half 10 of a shift clutch 10, 11. The other half 11 of the latter shift clutch is mounted fast on a transmission shaft 12, which also carries a wheel 17. The turbine wheel T of the converter carries a wheel 16 (on the engine side) of a second pair of wheels 15, 16, for transmission with the converter, with the interposition of a one-way clutch 18 (freewheel) and a hollow shaft 19. The co-acting wheel 15 is mounted on a countershaft 20. Keyed to the shaft 20 is a wheel 21 coacting with the wheel 8 of the main shaft train and in addition the halves 23 and 24 of each of two shift clutches 22, 23 and 24, 25. The shift clutch half 22 is combined with the co-acting wheel 2 of the first pair of wheels. The shift clutch half 25 is mounted on a countershaft 26, which carries keyed on it a wheel 27 coacting with the wheel 17 of the shaft 12 on the transmission side. The guide wheel L of the converter is mounted in the stationary housing with the interposition of an overrunning clutch 28 (free-wheel). In accordance with this embodiment of the propulsion plant with stepwise shifted gear in the multi-group type of construction, it is possible to accommodate the gear shift clutches 4, 5 and 22, 23 in such a manner as to save space, so that the gear is of particularly compact construction.

If it is desired to run without using the converter, the shift clutch 22, 23 is closed, so that the shaft 19 of the turbine wheel T is coupled to the engine or pump shaft M which carries the wheel 1. Power is therefore transmitted from the engine direct to both the clutch part 4 and to the countershaft 20, without using the torque converter. Since no counter-torque is then applied to the guide wheel L, the latter can freely turn in consequence of its mounting on the stationary housing, through the free-wheel 28. In the second and third gear speeds the hollow shaft 19 is turned more quickly, through the pair of wheels 8, 21, than the engine shaft, in consequence of the presence of the freewheel 18. The turbine wheel T is merely driven by the flow of fluid, while its speed is reduced and approximately equals the speed of the engine or pump wheel.

Figure 2 shows the tractive force diagram of a converter gear of this type. On the left, vertically, are indicated the values of the torque Mt of the transmission shaft, while horizontally are shown the speed values n_2 of the transmission shaft. Wi is the ideal line of tractive force of a converter with 100% efficiency, and IW and IIIW are its actual lines of tractive force when using the first and third gear speeds. The diagrams of the various gear speeds are designated by I, II, III, and IV.

In the embodiment illustrated in Figure 3, the turbine part T is mounted on a hollow shaft 31, with the interposition of an overrunning clutch 30. Keyed to the hollow shaft 31 is a gear wheel 32 and one part 33 of a double shift clutch 33, 34, 35. A wheel 36 coacting with the wheel 32 is mounted on a countershaft 37, which has also keyed to it a wheel 38 and one half 40 of a shift clutch 40, 41. The other half 41 of this shift clutch is fast on a shaft 43

together with a wheel 42. The centre part 34 of the double clutch 33, 34, 35 is secured on the engine shaft M, which carries the pump part P of the converter. The clutch part 35 is mounted, together with a wheel 45 and one half 46 of a clutch 46, 47, on a shaft 48. The clutch half 47 is keyed to the transmission shaft 49, which also carries a wheel 50. The construction illustrated in Figure 3 permits one pair of wheels to be saved. Here the engine or pump shaft M and the hollow shaft 31 are short-circuited by operating the clutch 33, 34. Otherwise, the mode of operation is exactly as in the case of the example illustrated in Figure 1. The guide wheel L of the converter is mounted in the stationary housing with the interposition of a free-wheel coupling 52.

Advantageously, certain structural alterations modifications may be made in the arrangement according to Fig. 1 in order to compensate for the slip which experience has shown is produced when driving the stepwise shifted gear through the converter, and as a consequence of which the maximum speed at any given moment of the turbine wheel does not quite reach the speed of the pump wheel under load. To this end, according to a feature of the invention this compensation is rendered possible by correction of the gear ratio in the transmission from the converter output to the gear input shaft, the gear wheels participating being of such dimensions that the difference between the gear ratios in the transmission from the turbine wheel to the stepwise shifted gear and in the transmission from the engine (pump wheel) to the stepwise shifted gear corresponds entirely or substantially to the above-mentioned slip between the turbine wheel at maximum speed and full load and the pump wheel. The effect is thus achieved that the maximum full load speeds of the output shaft of the gear when operating with the aid of the converter are equal to the speeds in operation when the converter is not used.

By bringing the converter into operation, the input torque of the stepwise shifted gear is brought to a multiple of the engine torque. In order then to avoid the larger and heavier construction corresponding to the higher torque, the stepwise shifted gear of the present invention is built as a so-called overdrive construction, the ratios in the stepwise shifted gear all being reduced by one stage jump, while in the transmission of torque to the vehicle wheel axle the ratio is increased by the above-mentioned stage jump. In the four-speed gear construction illustrated in Figure 1, for example, the values ϕ^{-1}, ϕ^{0}, ϕ^{2} would be provided instead of the values ϕ^{1}, ϕ^{0}, ϕ^{2} for the ratios of the three pairs of wheels to be driven direct by the engine. In order to prevent the pinion of the last pair of wheels (with the ratio ϕ^{2}) from receiving too high a speed of rotation, according to the invention the three values are so altered, while retaining the total ratios, such that the value for the last pair of wheels becomes smaller than ϕ^{2}; for example the values are fixed at $\phi^{-\frac{1}{2}}$, $\phi^{+\frac{1}{2}}$, $\phi^{+1\frac{1}{2}}$.

Figure 4 illustrates an embodiment of such an arrangement. It shows a four-speed gear, in which the gear wheels of the various gear stages appear in the correct dimensional proportions to one another.

The pump wheel P and the pinion 1 are driven by the engine (not shown), and the turbine wheel T is also driven from the pump wheel P through the guide wheel L. The turbine wheel T drives the pinion 16 through the free-wheel 18. At full load and maximum speed n_1 of the engine, the speed n_1 of the turbine wheel is smaller than the engine speed n_1 by a determined amount. Slip thus occurs. When the pinion 1 is driven directly, it runs at the engine speed n_1. The transmission ratios of the wheel pairs 1, 2 and 16, 15 are according to the invention so adjusted that the difference between n_1 and n_1 is balanced and the output speeds in the corresponding gears I without converter and I with converter, or III without converter and III with converter, are in each case equal.

With normal graduation of the individual gear stages I, II, III, IV, with the stage jump (greater than 1) and direct drive in the fourth speed, the graduation in the various gear speeds would be:—

First gear speed, transmission ratio $n_1/n_{I}=\phi^3$
Second gear speed, transmission ratio $n_1/n_{II}=\phi^2$
Third gear speed, transmission ratio $n_1/n_{III}=\phi$
Fourth gear speed, transmission ratio $n_1/n_{IV}=\phi^0=1$.

Thus the individual wheel pairs have the transmission ratios:—ϕ^0 and ϕ^2.

According to the invention the stage jump should for example be reduced by ϕ in each gear speed, so that for the individual force paths diagrammatically

illustrated in Fig. 4 the following transmission ratios are obtained:—

First gear speed: $n_1/n_I = \phi^2$
Second gear speed: $n_1/n_{II} = \phi$
Third gear speed: $n_1/n_{III} = \phi^0 = 1$
Fourth gear speed: $n_1/n_{IV} = \phi^{-1} = 1/\phi$

Hence the direct drive thus occurs in the third gear speed. The fourth gear speed gives a step-up ratio. By increasing the ratio at the wheel axle of the vehicle by the stage jump, the original transmission ratios ϕ^3, ϕ^2, ϕ, and ϕ^0 are once again obtained.

In order to avoid too high speeds of the pinion 27 in the fourth gear speed, the exponent of the stage jump for the wheel pair 17, 27 is reduced in relation to the exponent of the maximum transmission stage of the gear, for example to $1\frac{1}{2}$. In order to retain the above-indicated total ratios in the individual gear stages, the wheel pairs of the stepwise shifted gear are constructed with the following transmission ratios in the exemplified embodiment illustrated:—

	Transmission Ratio
Wheel pair 1, 2	$\phi^{-\frac{1}{2}}$
Wheel pair 8, 21	$\phi^{-\frac{1}{2}}$ or $\phi^{+\frac{1}{2}}$
Wheel pair 17, 27	$\phi^{+1\frac{1}{2}}$

The gear speeds of the stepwise shifted gear are then disposed as follows:

Gear speed	Wheel pairs	Clutches	Stage jump of wheel pairs	Total transmission ratio
I	8, 21; 27, 17	4 and 25	$\phi^{+\frac{1}{2}}$ $\phi^{+1\frac{1}{2}}$	ϕ^2
II	1, 2; 27, 17	22 and 25	$\phi^{-\frac{1}{2}}$ $\phi^{1\frac{1}{2}}$	ϕ^1
III	direct	4 and 10		ϕ^0
IV	1, 2; 21, 8	22 and 10	$\phi^{-\frac{1}{2}}$ $\phi^{-\frac{1}{2}}$	ϕ^{-1}

Thus with the reduced exponents the same total transmission ratios are obtained in the individual gear speeds, as with the original exponents. The wheel pair 8, 21 is used twice, namely in the first gear speed with the stage jump $\phi^{+\frac{1}{2}}$ and in the fourth gear speed with the stage jump $\phi^{-\frac{1}{2}}$.

What we claim is:—

1. A power transmission device for motor vehicles, comprising a hydraulic transmission (torque converter) combined with a spur wheel change-speed gear equipped for more than two forward gear speeds and having coaxial main shafts and a countershaft parallel thereto, wherein the shaft connected to the turbine wheel of the converter is constructed as a hollow shaft and surrounds the gear input shaft, and non-positive gear shift clutches incorporated between the wheel pairs for the purpose of producing the different speeds both in individual gear stages with the use of the hydraulic transmission and in individual gear stages of the stepwise shifted gear alone while the hydraulic transmission is disconnected, characterised in that the gear input shaft and the turbine wheel hollow shaft are each in operative connection with one of the non-positive gear shift clutches situated inside the gear, and through the operation of which the gear input shaft is coupled in each case to the turbine wheel shaft and thus only gear shift clutches are used for connecting and disconnecting the hydraulic transmission, without additional clutches being required.

2. Device as claimed in Claim 1, wherein one gear shift clutch of the countershaft is provided for establishing connection between the pump wheel shaft and the turbine shaft, the two clutch halves being each connected through a pair of gear wheels to the gear input shaft and to the turbine shaft, respectively.

3. Device as claimed in Claim 1, wherein one gear shift clutch of the main gear shaft is provided for establishing connection between the pump wheel shaft and the turbine shaft.

4. Device as claimed in Claims 1 and 2, wherein for the purpose of compensating for speed slip between the turbine wheel and the pump wheel of the converter in the stepped gear, the transmitting gearwheels are of such dimensions that the difference between the gear

ratios in the transmission of power from the turbine wheel to the stepwise shifted gear and in the transmission of power from the pump wheel (engine) to the stepwise shifted gear corresponds entirely or approximately to the above-mentioned difference in speed (slip) between the turbine wheel and the pump wheel.

5. Device as claimed in any of Claims 1 to 4, wherein for the purpose of reducing the gear load and hence reducing the over-all dimensions, the stepwise shifted gear is constructed as a group gear of the overdrive type.

6. Device as claimed in any of Claims 1 to 5, characterised in that, in order to avoid too high speeds of the smallest countershaft wheel the exponent of the ratio of the pair of wheels enclosing this countershaft wheel is reduced, while retaining the gear ratios, in relation to the exponent of the maximum ratio of the stepwise shifted gear.

7. Device as claimed in Claim 6, having a stepwise shifted gear of the overdrive type, with three pairs of wheels, wherein while retaining the overall ratios, the ratio for the last pair of wheels is selected to be smaller than ϕ^2 and for example the values $\phi^{-\frac{1}{2}}$, $\phi^{\mp\frac{1}{2}}$, $\phi^{+1\frac{1}{2}}$ are fixed for the three pairs of wheels, so that in the case of the exponents of the first and second values the total equals -1, and in the case of the exponents of the first and third values the total $=1$, while the total of the positive exponent of the second pair of wheels and the exponent of the third pair of wheels $=2$.

8. The power transmission device for motor vehicles constructed, arranged and adapted to operate, substantially as herein described and with reference to the accompanying drawings.

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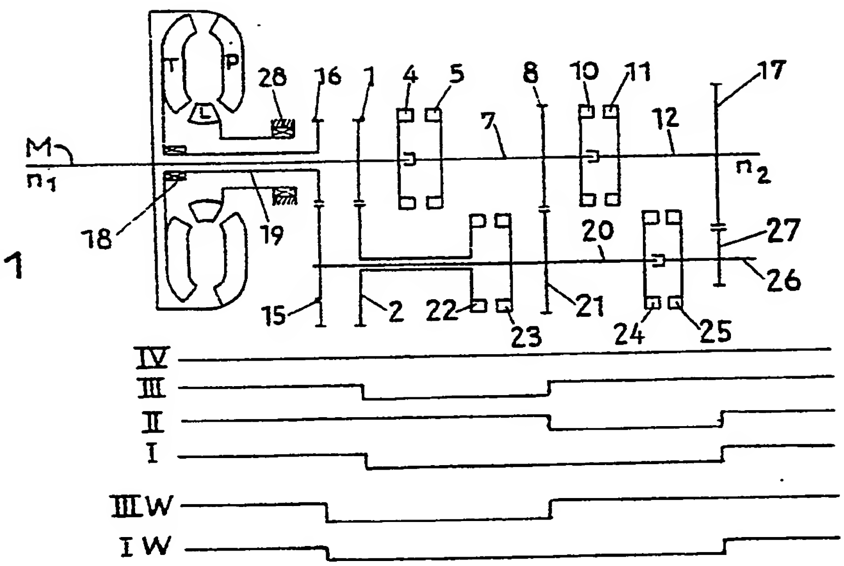
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Fig. 1

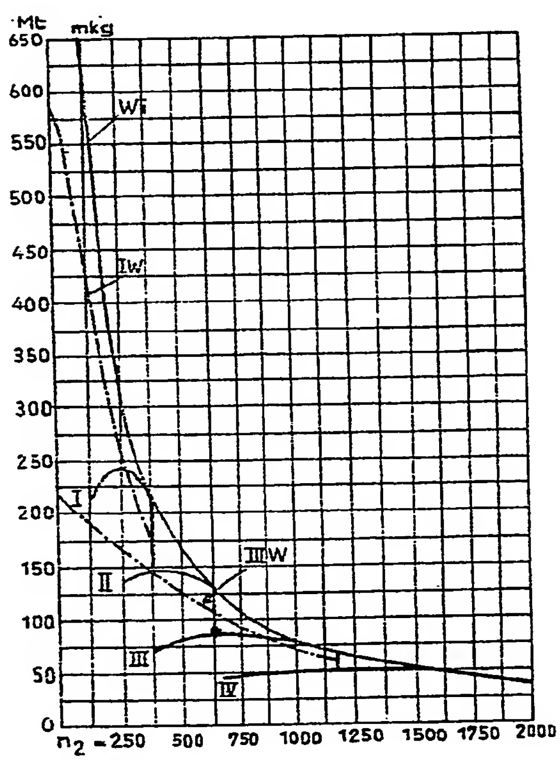
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Fig. 2

721,365 COMPLETE SPECIFICATION
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SHEETS 1,2 & 3

Fig.3

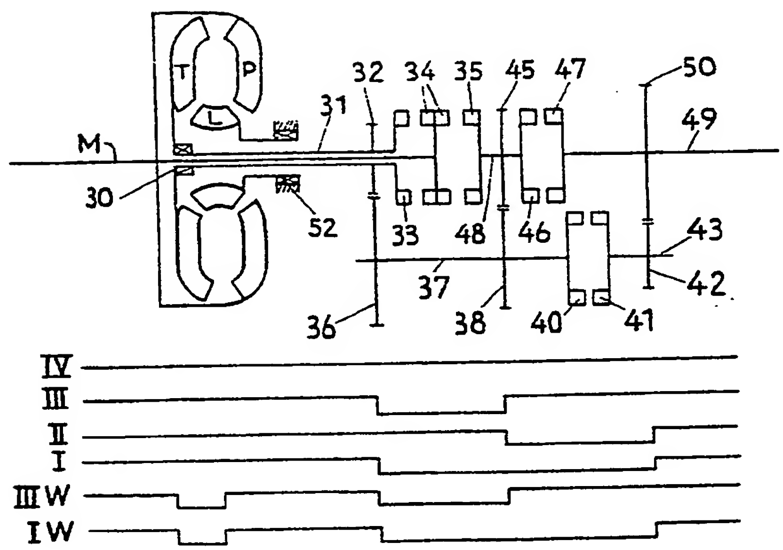

Fig.4.

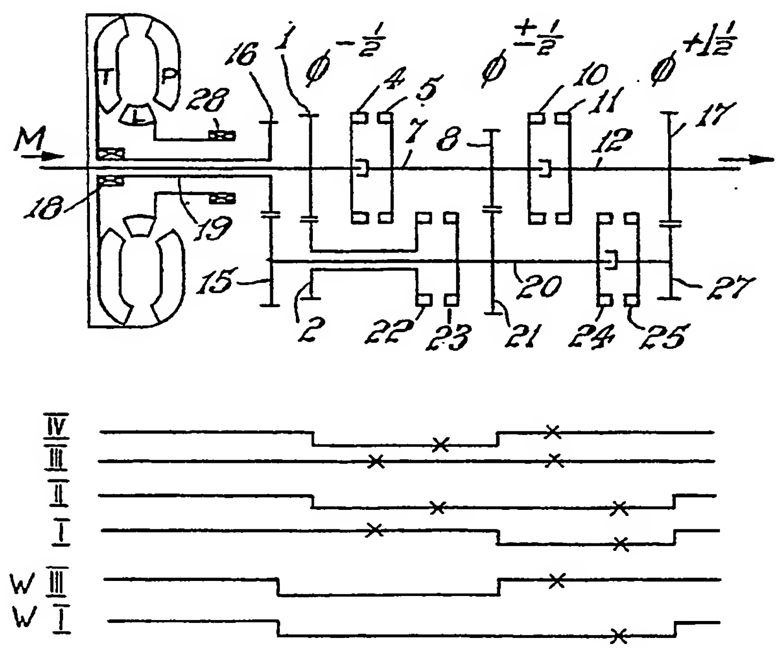

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Fig. 1

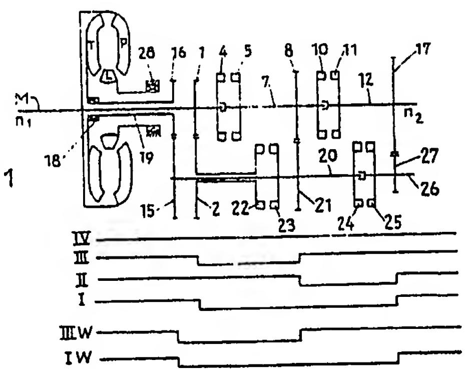

Fig. 2

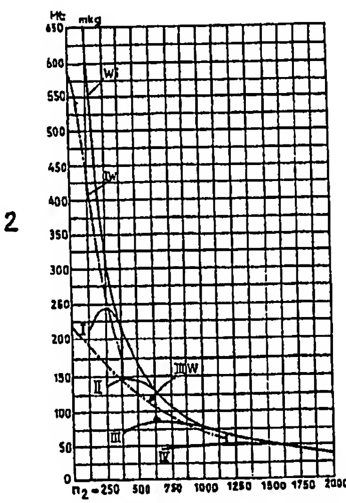

Fig. 3

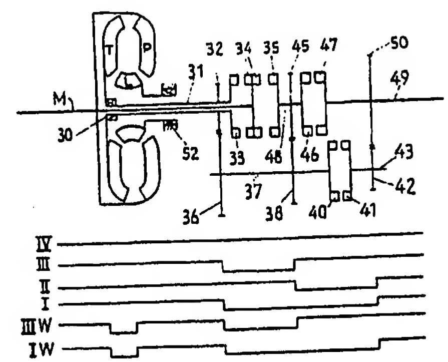

Fig. 4.